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Mars Sample Return Power Supply

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i. ABSTRACT

Purpose of this design is to provide a power supply for a vehicle which will be able to operate on Mars for a time period of about five to ten years. This vehicle will be used for sample and data collection on the Martian surface. This design will be based on the assumption that the vehicle will be unmanned. Also, there will not be any means by which components could be repaired or replaced while on the Martian surface. A consequence of this drawback is that all equipment will be forced to meet a high standard of reliabilty and, if possible, redundancy.

Power will be supplied to the vehicle by means of a General Purpose Heat Source capable of producing 7kW of minimal thermal power. The heat generated from the General Purpose Heat Source will transferred to a Stirling engine via "hot side" heat pipes. The Stirling engine will then convert this heat into 2 kw electrical power. "Cold side " heat pipes will be used to carry away waste heat from the cold side of the Stirling engine. This heat will then be released to Martian atmoshere via radiators connected to the end of the "cold side" heat pipes.

Attributions

Don Hoang	: cold side heat pipes and radiator
Sharon Ludwigs	: hot side heat pipes, introduction, abstract
Paul Schmitz	: stirling engine/linear alternator, mass optimization
John Wright	: heat source, results/conclusion, summary

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ii. INTRODUCTION

For centuries scientist have studied the planet of Mars from afar by means of telescopes, satelites, and deep space probes. Much effort has been put into projects which would allow for actual study of the Martian geology, chemistry, and other fascinating information. This information could be gathered by sample and data collection on the Martian surface.

The object of this design is to conceptually develope a vehicle for this purpose. This project will be appropriately called the "Mars Sample Return Mission." The vehicle will be required to be light weight, reliable, and if possible, to have redundant parts in case of a component failure.

This study will be restricted to the power supply of the vehicle; equipment necessary for the actual collecting of the soil samples will not be included in this study. The design will be divided into four categories consisting of the power supply, power converter, heat transfer devises, and a radiator. A mass optimization will then be performed on the entire system and the system efficiency calculated.

I. The Heat Source

I. The Heat Source

Heat generation for the power supply in this design is accomplished with a general purpose heat source (GPHS), which is similiar in design to others used in satellite radioisotope thermoelectric generators.

The fuel is PuO_2, composed of the plutonium-238 isotope. This fuel form is used almost exclusively in radioisotope heat sources. This plutonium isotope has a long half-life, 87.74 years, which gives it a reasonably constant power generation rate over the mission lifetime. The mode of decay is the emission of a 5.5 MeV alpha particle, giving a power density of 4.96 W/cm^3.

A GPHS module is illustrated in Figure 1-1. Each module contains four cylindrical PuO_2 pellets encased in iridium cladding. These are in turn encased in graphite cylinders, which are part of the graphite shell. Iridium was chosen because 1) it has a high melting point, and 2) it has been used previously in this type of heat source. Graphite was chosen for the same reasons. The fuel pellets are 2.522 cm in length with an equal diameter. The iridium cladding is 0.01 cm thick. Overall module dimensions are 10.16 cm x 10.16 cm x 5.08 cm. The mass breakdown is 0.508 Kg of PuO_2, 0.027 Kg of iridium, and 0.758 of Kg graphite for a total module mass of 1.293 Kg. Both the iridium clad and the graphite cylinders are vented to allow the release of the helium produced in the decay process. The clad and the graphite also serve to contain the fuel in case of an accident. Each module produces a power of 250 W.

General Purpose Heat Source

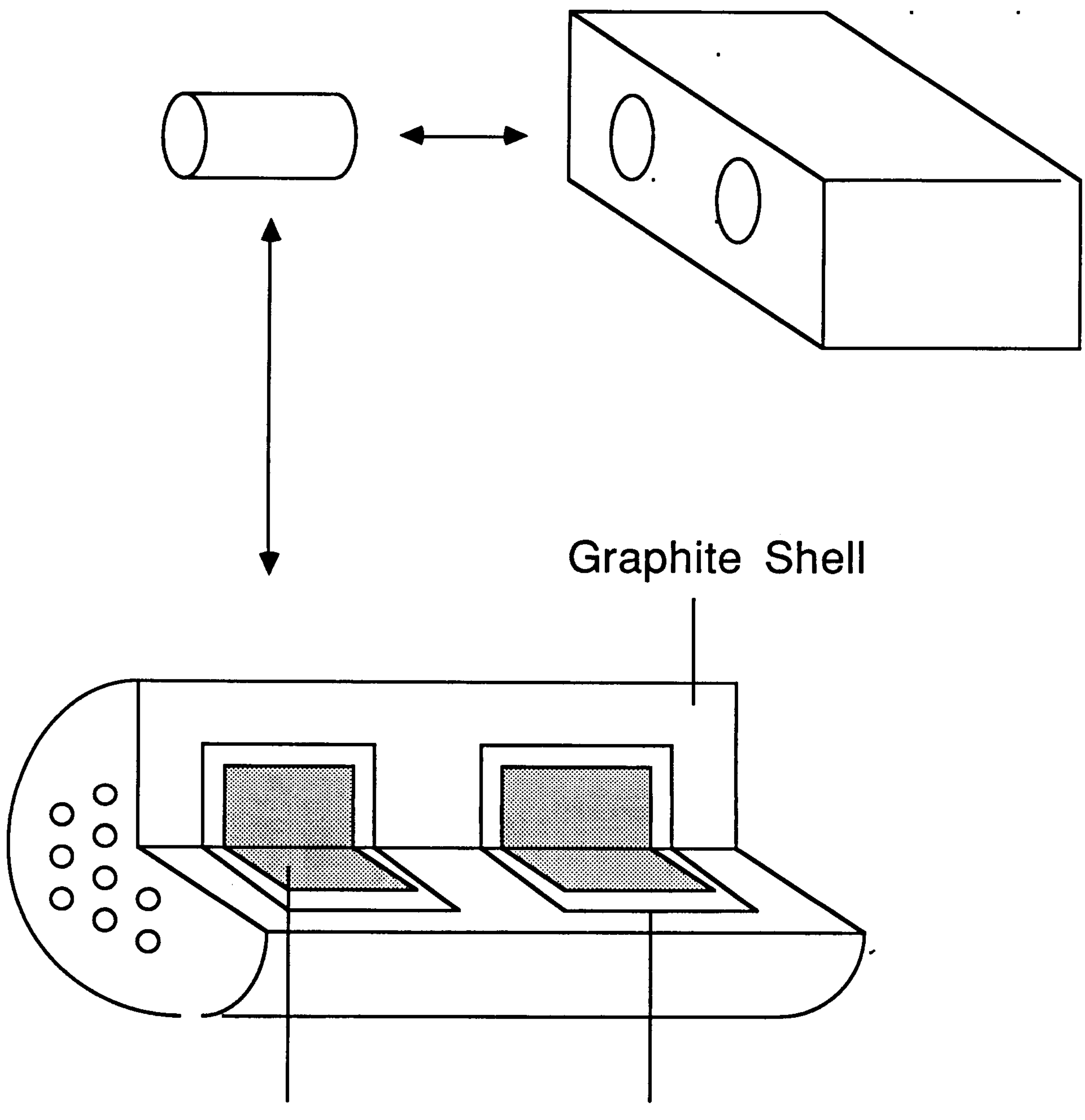


Figure 1-1

Each GPHS assembly consists of ten stacked modules, which are encircled by heat pipes, and enclosed in 0.5 cm thick titanium casing. This is shown in Figure 1-2. Casing dimensions are 51 cm high and 20 cm diameter. The casing mass is 8.638 Kg. This gives a total assembly mass of 21.568 Kg, excluding the heat pipes, which are attached to the sides of the module stack.

The entire heat source consists of three GPHS assemblies, with a total mass of 64.704 Kg. This is 20-25 Kg lighter than similiar heat sources used in Brayton and Dynamic Isotope power systems. The weight savings is due to the use of titanium instead of stainless steel as the casing material. The heat source has an initial power of 7.5 Kw thermal, which declines to 6.92 Kw thermal after ten years due to the plutonium decay. This is adaquate to supply the 2 Kw-electric mission requirement.

Figure 1-3 illustrates the model used for the temperature profile. The following four assumptions were made:

1) the fuel pellets are placed symmetrically within each module, giving an equal temperature distribution on each side,
2) the cylindrical pellets can be approximated as cubes of equal volume,
3) gaps between the fuel, cladding, and graphite are negligible,
4) from any side, the heat source can be considered as a fuel rod whose diameter is equal to twice the cube width.

The temperature profile was found with the following temperature distribution equation for slab geometry:

GPHS Assembly

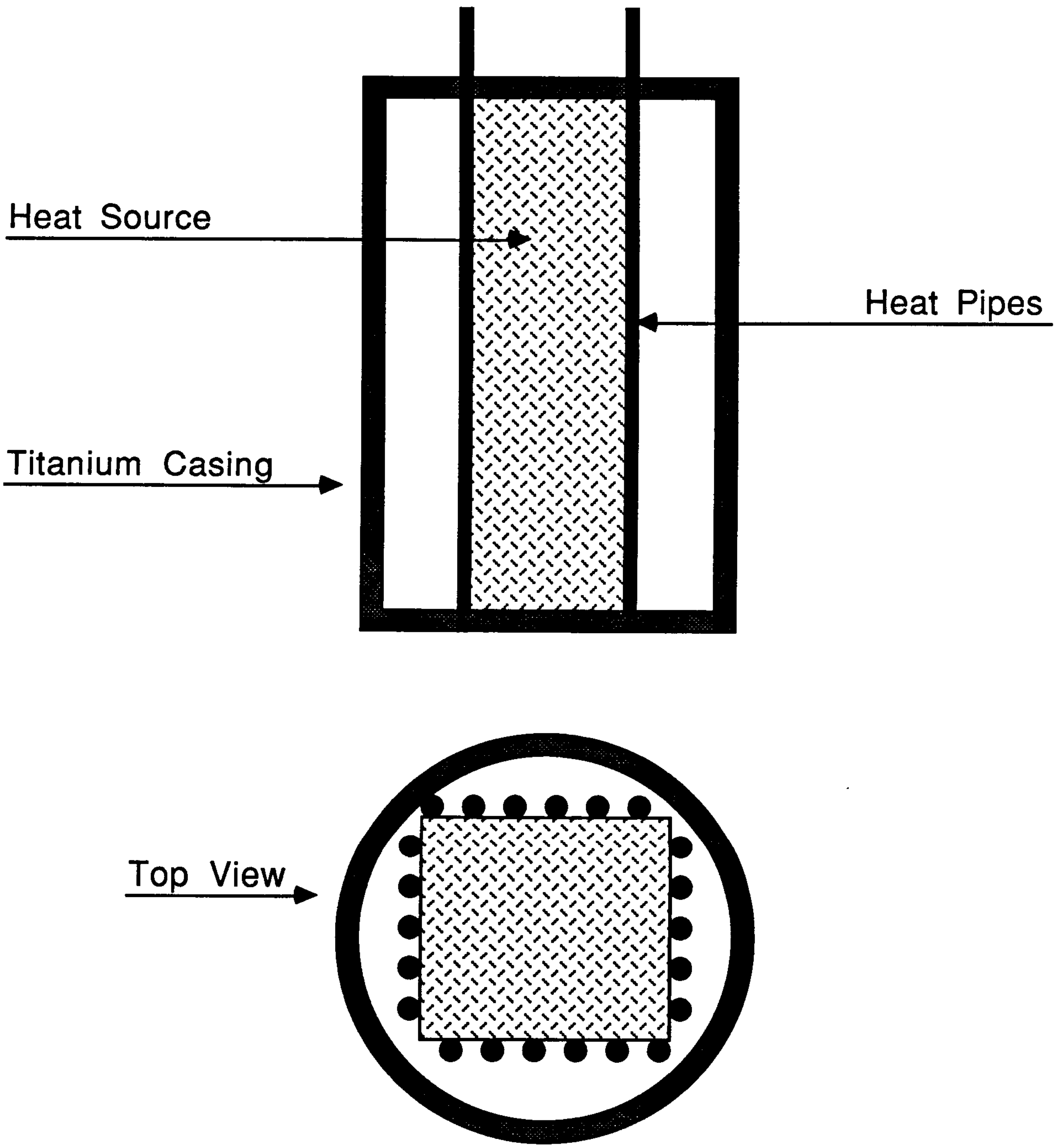


Figure 1-2

GPHS Temp. Model

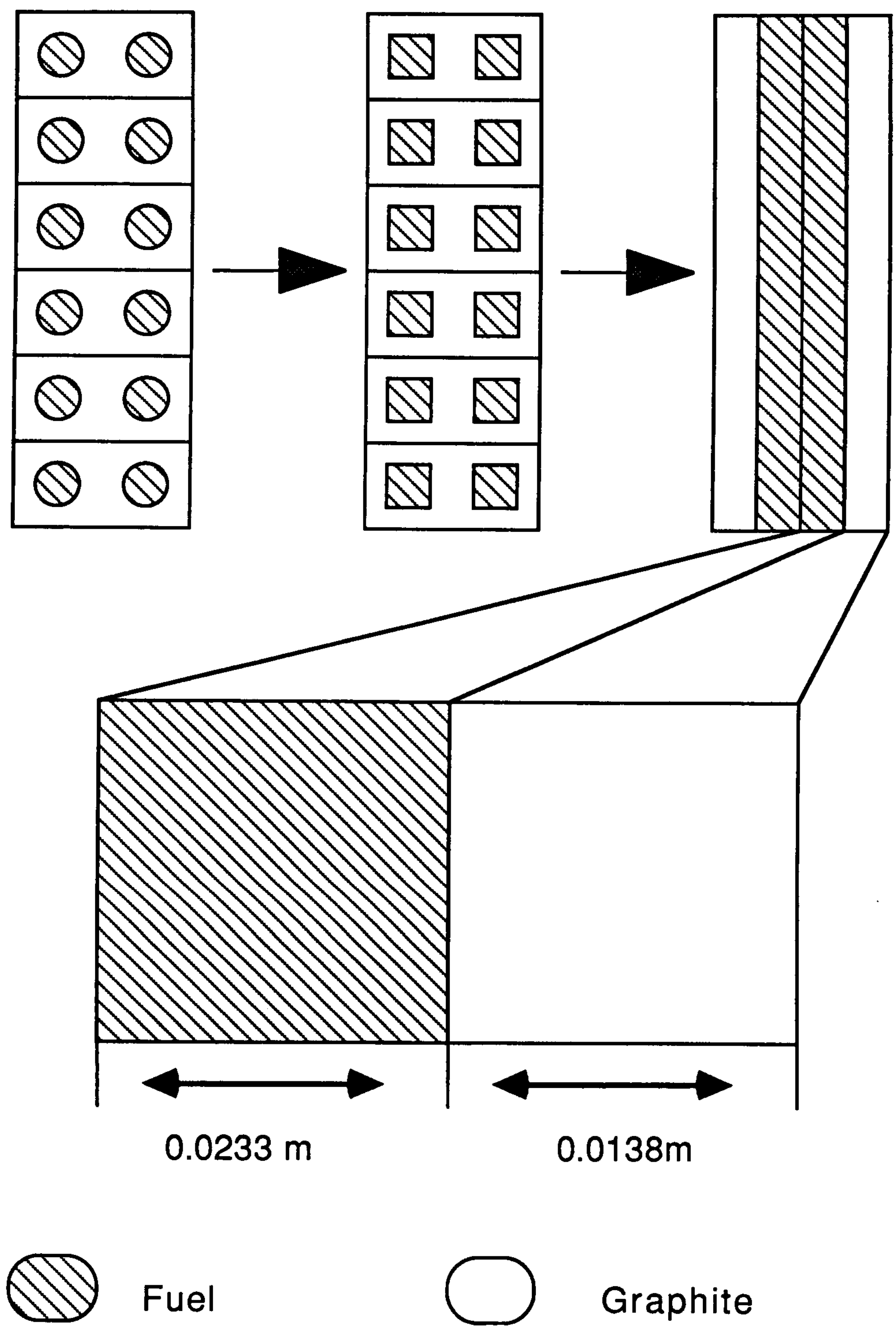


Figure 1-3

$$T\text{-}T_{wall} = q'' \left[\left(X_f^2 - X^2 \right) / \left(2 k_f X_f \right) - X_c / k_c \right] \quad \text{Eq.(1-1)}$$

$$\text{and} \quad q'' = q''' \; X_f, \qquad \text{Eq.(1-2)}$$

where $T\text{-}T_{wall}$ is the temperature difference between the fuel rod center and the outer graphite wall (K), q''' is the power density (W/m^3), q" is the heat flux (W/m^2), X_f is the fuel rod radius (m), k_f is the fuel thermal conductivity (W/m-K), X_c is the graphite thickness , k_c is the graphite thermal conductivity, and X is the distance from the fuel rod center. This model gives a maximum temperature difference of 771K (498C). Maintaining the wall at 1000K yields a fuel centerline temperature of 1771K (1498C). Figures 4 and 5 show the profiles. These temperatures are well below the melting point of the materials, 2870K for PuO_2, 2725K for iridium, and 3975K for graphite. Note that the iridium clad was ignored for the profile calculation. This is because $X_{iridium}/k_{iridium} << 1$, and changes the temperature difference by less than 1K.

Consideration was also given to the buildup of helium gas over the mission lifetime. It was assumed that all the helium was vented through the cladding and graphite shell into the casing. First, the total power produced by 1 cm^3 over the mission lifetime was calculated from the equation (1-3):

$$P_{tot} = \int P_o e^{-t/\tau} dt = P_o \tau (1 - e^{-t/\tau}),$$

where P_{tot} is the total power , P_0 is the power density, τ is (half-life) / (ln2), and t is the elapsed time. Over ten years the power produced is $1.5x10^9$ Ws/cm^3. With 1 Ws = $1.135x10^{12}$ helium atoms, giving a total helium output of 0.0028 moles / cm^3 produced. The module fuel volume is 50.379 cm3, giving 0.142 moles helium / module. With ten modules in each heat assembly, there is 1.42 moles of helium produced. Solving the Ideal Gas Law for pressure ,

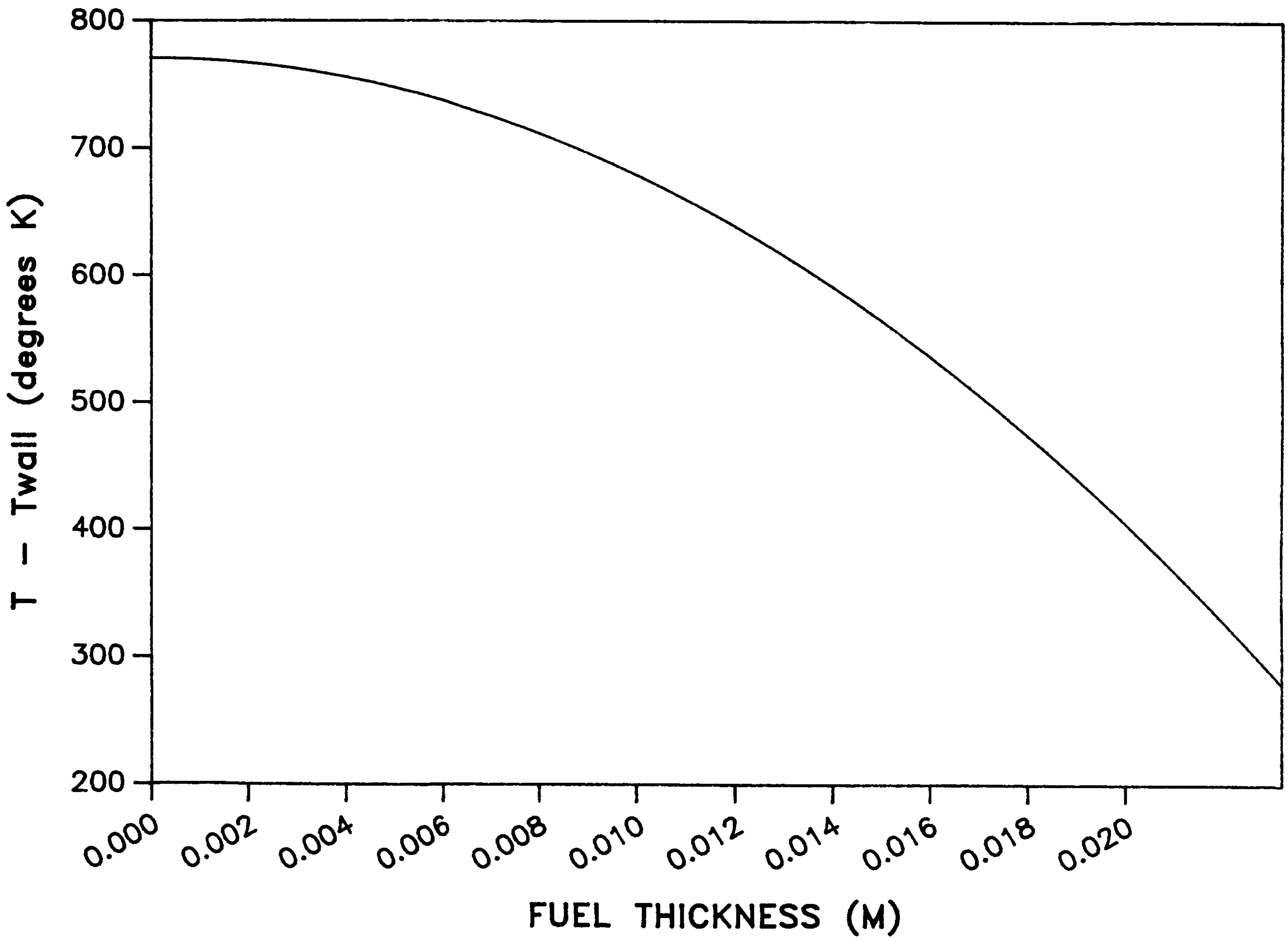
$$P = (nRT / V), \qquad \text{Eq.(1-4)}$$

where n is the number of moles of gas (1.42), R is the gas constant (8.314 KJ/Kg-mole-K), T is the temperature (1000K), and V is the casing air volume (0.0107 m^3), yields a pressure buildup of 1.1 MPa, or 160 psi. This pressure can easily be contained in the titanium casing.

The GPHS overall characteristics are listed in the summary.

Figure 1-4

GPHS TEMP. PROFILE



GPHS TEMP. PROFILE

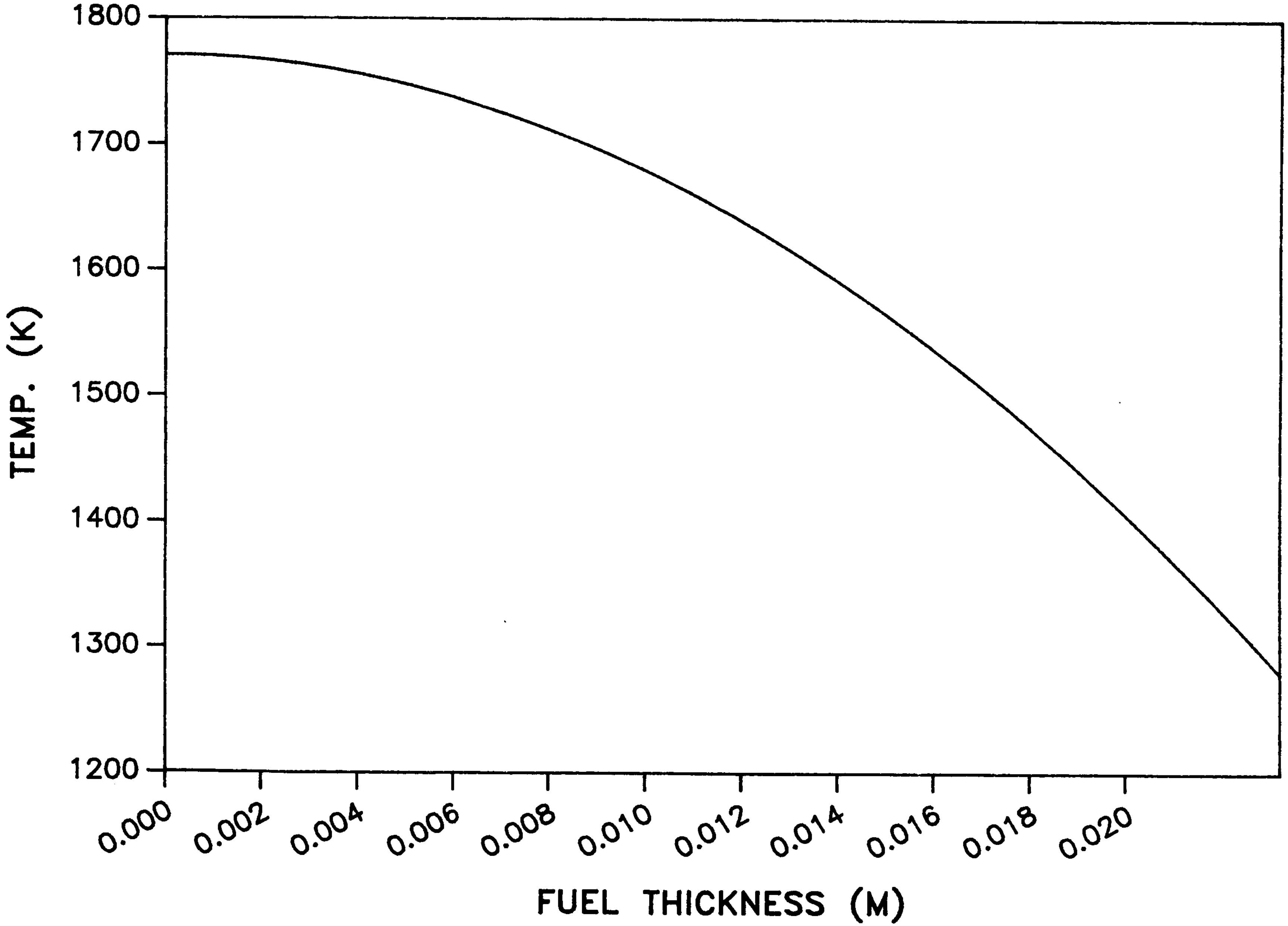


Figure 15

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II. Stirling Engine

INTRODUCTION

Developement of Stirling engines is proceeding in spite of their admittedly higher cost because of their high efficiency, their ability to use any source of heat, there low vibrational mode of operation and their long life.[1] Stirling engines have until recently not been considered for space operations due to problems with producing large amounts of power. For the Mars Sample Return Mission(MSRM) the Stirling engine is ideal for the lower power levels at which it will be operating. It is also much more reliable than a turbine powered vehicle due to its piston configuration rather than weak turbine blades. Cost considerations for space missions are generally only paramount when considering lauch costs. The price of developement of the Stirling engine would be relatively low for the MSRM and NASA is already working on a similiar engine for space applications.

Theory

Like any heat engine, the Stirling engine goes through four basic processes of compression, heating, expansion, and cooling. In a Stirling engine, these processes occur sequentially but partially overlapping in time. In different parts of the machine ,however, the boundaries of these processes are not definite. For this discussion a heat engine is a Stirling engine when:

1. The working fluid is contained in one body at nearly a common pressure at each instant during the cycle
2. The working fluid is manipulated so that it is generally compressed in the colder portion of the engine and expanded

generally in the hot portion of the engine.

3. Transfer of the compressed fluid from the cold to the hot portion of the engine is done by manipulating the fluid boundaries without valves or real pumps. Transfer of the expanded hot fluid back to the cold portion of the engine is done the same way.
4. A reversing flow regeneration is used to increase efficiency.[2]

Figure 2-1 shows the general process which converts heat into mechanical energy. A hot and cold gas space is connected by a gas heater, cooler, and regenerator. The working fluid is compressed in the cold space, transferred as a compressed fluid into the hot space where it is expanded again, and then transferred back again to the cold space. Net work is generated during each cycle and is equal to the area of the enclosed circle.

The thermodynamic definition of a Stirling cycle is isothermal compression and expansion at constant volume heating and cooling shown in Figure 2-2. One of the requirments for Stirling engines is to have high efficiency regenerators. Now let us consider a annular gap around the displacer which acts as a gas heater, regenerator and cooler (see Figure 2-3). Let E be the regenerator efficitiveness during the energy transfer. The following symbols represent transfer from cold space to hot space:

Let TL = temperature of gas leaving regenerator

TC = TC(N) for any N

TH = TH(N) for any N where N is position

2-1.) E = (TL-TC)/(TH-TC)

During transfer the heat from the regnerator is :

Figure 2-1

The Stirling Cycle:
A heat power cycle using isothermal compression and expansion, and constant volue heating and cooling

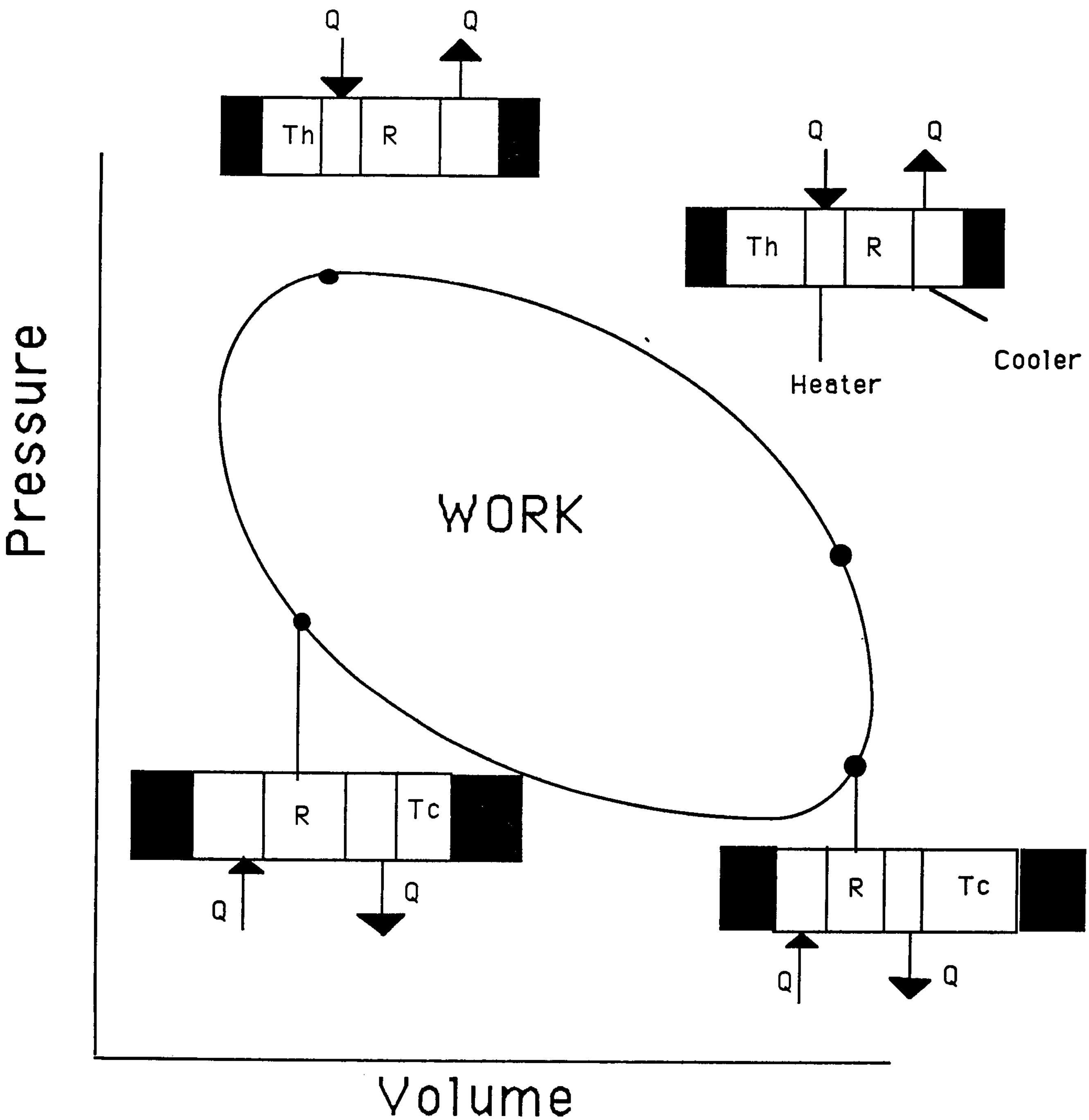
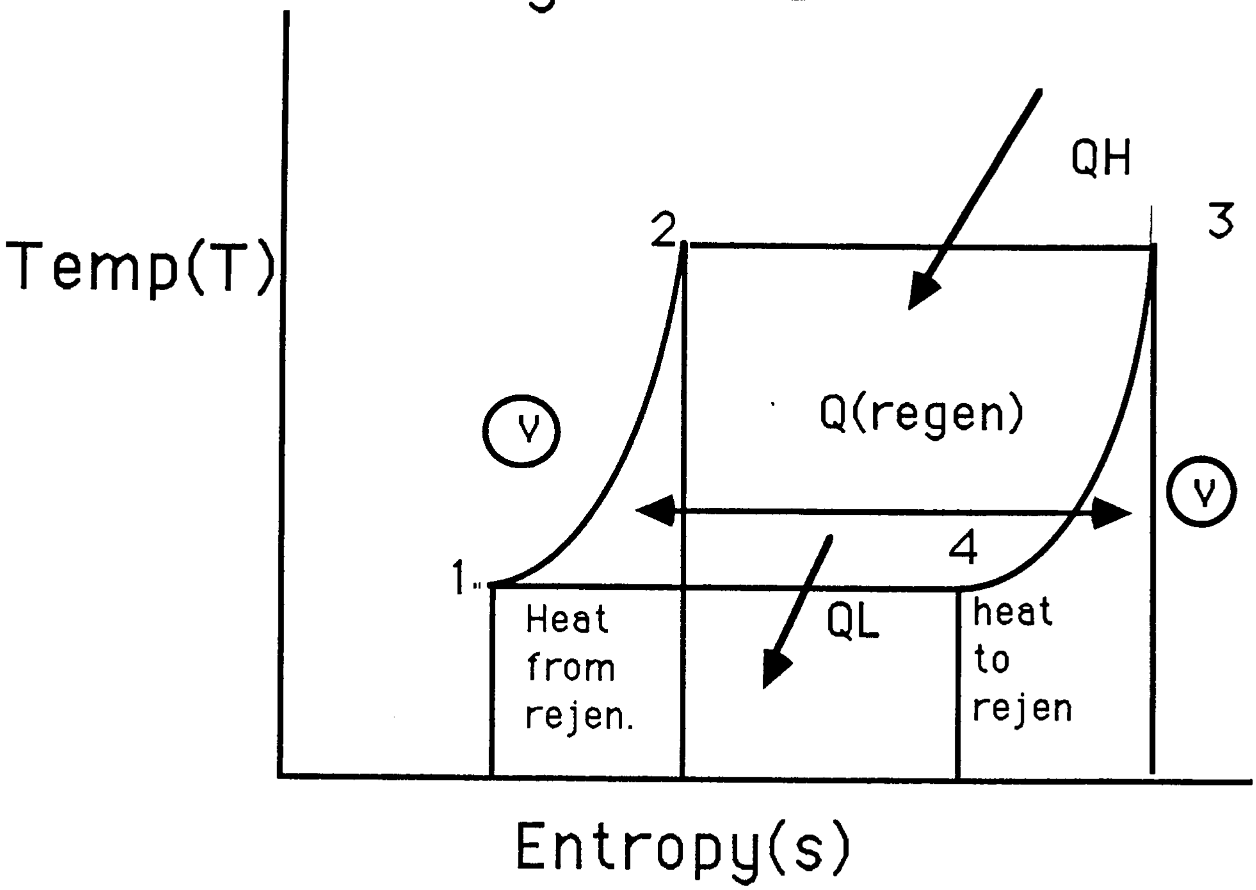


Figure 2-2



V Means Constant Volume Process

1-2 Constant Volume Process

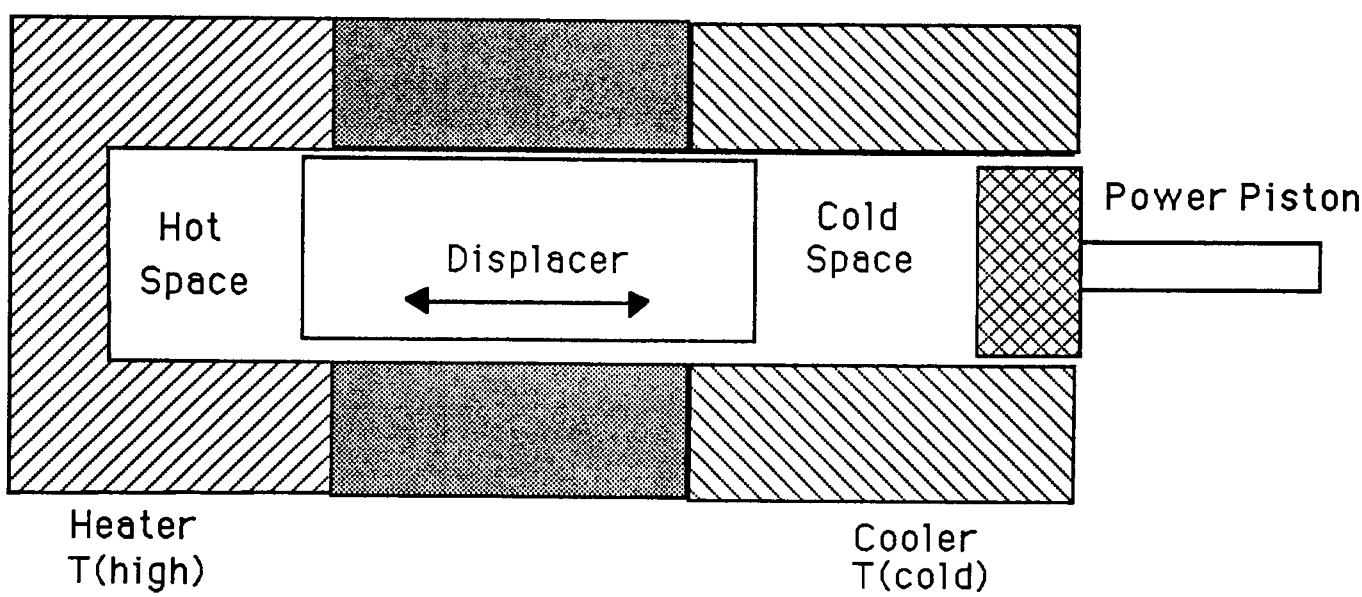
2-3 Isothermal Expansion

3-4 Constant Volume Cooling

4-1 Isothermal Compression

Figure 2-3

Simple Stirling Engine with Annular Gap Regenerator



2-2.) $QR = M(CV)(TL - TC)$

and the heat from the gas heater is :

2-3.) $QB = M(CV)(TH - TC)$

So the efficiency becomes:

2-4.) $EF = (TH - TC)/(TH + CV/R\ [(TH-TC)(1-E)/(\ln(V(1)/V(2))]$

This equation relates the hot and cold temperatures and the volume ratio to the efficiency of the engine.[3] Figure 2-4 shows how important this is for a volume ratio of 2, with a working fluid that is Helium, and a hot and cold side temperature of 1000K and 500K respectively.

Efficiency of a Stirling engine is related to the Carnot efficiency, which is directly related to the hot and cold side temperatures of the Stirling engine. The following formula for well optimized engines operating with helium gives a relation between Carnot efficieny and and η.

2–5.) $\eta_{eff} = P_{net}/E_f = (1 - T_C/T_H) * C * \eta_H * \eta_M * f_A$

where

η_{eff} = overall thermal effective efficiency

P_{net} = net shaft power with all auxiliaries driven

Ef = fuel energy flow

T_C, T_H = compression and expansion gas temperature, K

C = Carnot efficiency ratio of indicated efficiency to Carnot efficiency, normally from .65 to .75.

η_H = heater efficiency, ratio between the energy flow to the heater and the fuel energy flow. Normally between .85 and .90

η_H = mechanical efficiency, ratio of indicated to brake power, around .90

f_A = auxiliary ratio. At maximum efficiency f_A = .85

Thus the most optimistic figures for Stirling cycle efficiency is

2–6.) $\eta_{eff} = (1\text{-}TC/TH)(.75)(.90)(.90)(.95) = (1\text{-}TC/TH)(.58)$.[4]

Some attempts have been made to relate the power actually realized in a Stirling engine to the power calculated from the dimensions and operating conditions of the engine. The power output of many Stirling engine conform to

2-7.) $P = .015\, p \cdot f \cdot V_o$

where

P = engine power, watts

p = mean cycle pressure, bar

f = cycle frequency of engine speed, hertz

V_o = displacement of power piston, cm^3.

<u>Martian Sample Return Stirling Engine</u>

The configuration chosen for the Mars Sample Return mission heat conversion unit is that of a Stirling engine. It produces 2 kW of electrical power for use by the rover for a period of 10 years. Extremely high reliability was sought in the design since it was not forseen as being able to be repaired if a mechanical failure occurs. Stirling engines also offer the highest reliability of all known heat engines which directly affects radiator size. A Stirling engine was chosen over an RTG configuration because of the RTG's extremely low efficiency. Brayton and Rakine turbine cycles were also looked at, but because of possible turbine breakage when moving over rocky terrain it was decided that these systems would not offer the reliability sought. A single piston stirling engine with a linear alternator was therefore chosen for the generation of power. This configuration has only two moving parts, the dispacer and the power piston, and is hermetically sealed from the enviornment. A hot side temperature for the Stirling engine was set at 1000K by material constraints on the general purpose heat source. Figure 2-5 shows a outline of the engine.

Beryllium was chosen as the Stirling engine and linear alternator housing. Beryllium was chosen over stainless steel and aluminum which had been used in previous designs due to its extremely low weight, 1.85 g/cm^3 , and high thermal conductivity, 2 W/mK,. Beryllium also melts at 1560K which is well above the heater head temperature of 1000K. Beryllium is also highly magnetically permeable which allows excellent compatibility with the linear alternator assembly.

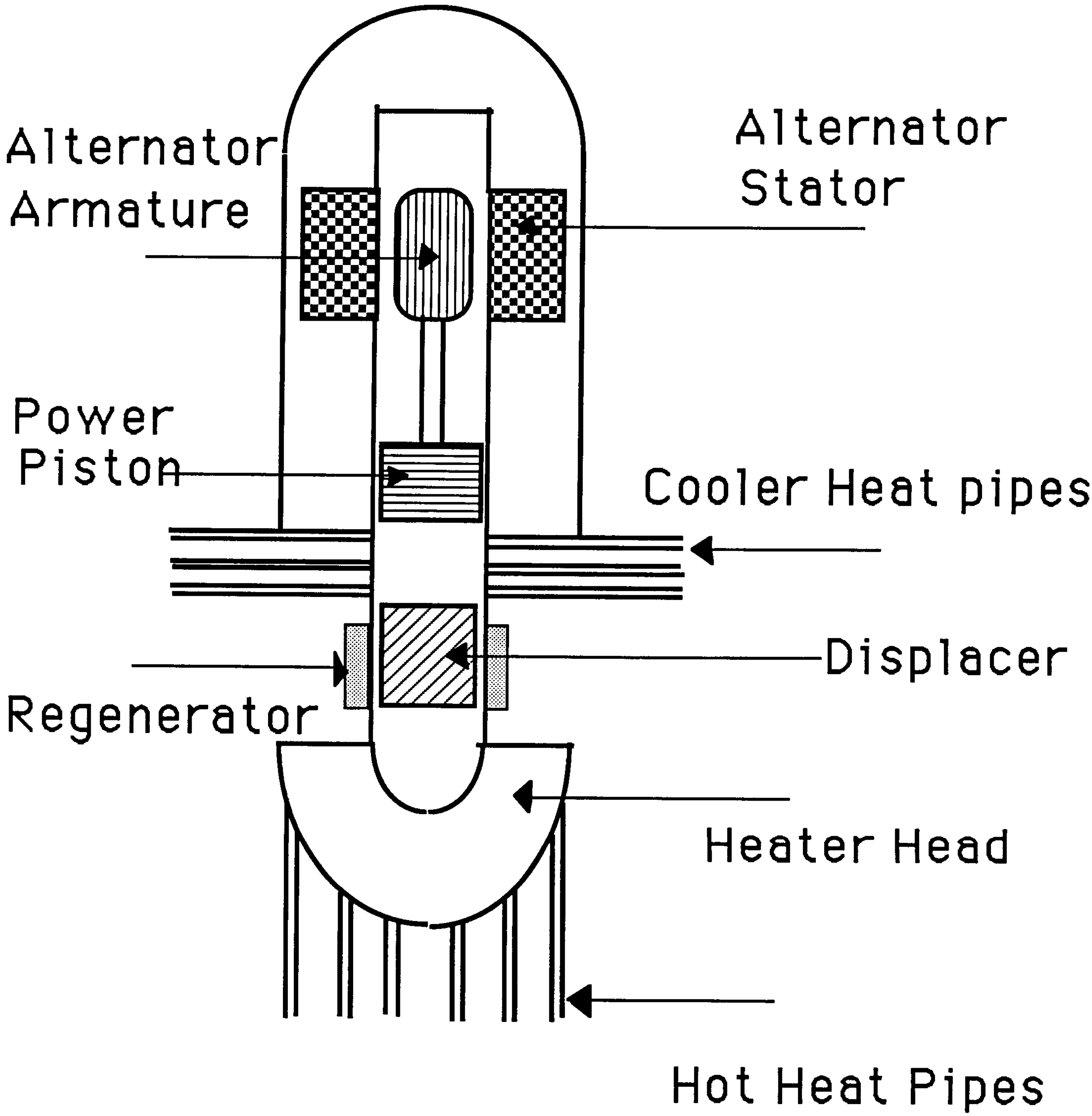
The working gas chosen was helium due to its high thermal conductivity for a gas, .33 W/mk, and proven use in past Stirling engines.[5] Other gasses that were looked at were nitrogen and oxygen but these were dismissed due to their lower efficiencies. Further study may show it is possible to recharge the Stirling engine if a leak should develope with the helium given of by the GPHS.

The power piston has a dispacement of 93.3 cm^3 and a frequency of 40 Hz. This cycle speed was chosen for its reliability. By use of equation 2-7 the dispacement was found. Forty and Seventy bar pressure charges were compared. The forty bar presssure charge was found to be less likely to cause material strengh problems as well as enabiling the walls of the Stirling engine to be thinner. Thus the forty bar pressure charge will be used as an operating parameter for the stirling engine.

The cold side temperature was found by performing a mass optimization for the entire system. It was found that a temperature of 498 K would give the minimum mass for this system. This cold end temperature gives an efficiency of 29.6% found by using equation 2-6. Then, knowing engine efficiency, the required effectiveness the rejenerator can be found which will allow the engine to function correctly. Figure 2-4 generated by the use of equation 2-4 shows regenerator effectiveness. The graph shows engine efficiency increasing with increasing regenerator effectiveness . In reality due to increased viscous losses of the working gas through the regenerator ,because of increased porosity of the regenerator, engine efficiency actually goes down as regenerator effectiveness approaches 1. For the engine, in this design, a regnerator efficiency of 62% was found. This value which is lower than the limit of 93% for decreased engine efficiency.[6]

The linear alternator assembly consists of a permanant magnet connected to the power piston. As the power piston moves the attached magnet is pushed and pulled through the alternator stator or coils which in turn create the current. Linear alternator efficiencies corresponding to power in the 2 kW range are around 90%.

Figure 2-5


Alternator Armature
Alternator Stator
Power Piston
Cooler Heat pipes
Displacer
Regenerator
Heater Head
Hot Heat Pipes

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III. Heat Pipes & Radiator

III. HEAT PIPES

For space applications heat pipes are an effective means of transferring heat. A heat pipe is a self-contained system which boils fluid at one end, called the evaporator, and condenses the vapor at the other end, which is called the condensor. Usually, there is a section between the evaporator and condensor which is called the adiabatic section. The main driving force in heat pipes is capillary action. It is this force that returns the condensed fluid is back to the evaporator. The heat source is located at or near the evaporator section and the heat sink is located at or near the condenser section. The liquid flow is through a wick structure which lines the inside of the pipe. The primary ways that the transfer of heat to and from the heat pipe takes place is: by conduction from a heat sink, by convection, or by thermal radiation. Thermal conduction through the heat pipe at both the evaporator and condenser walls will cause temperature drops to occur. If the vapor pressure gradient is kept small, the axial temperature gradient is also kept small resulting in a highly thermal conductivity. The effective thermal conductivity of the device can be more than 1000 times greater than a rod made of copper of similar dimensions.[1] Another difference between solids is that the performance of heat pipes is dependent upon the properties of the working fluid, heat transfer rate, size, shape, and the kind of containment and wick materials being used. Figure 3.1 illustrates a conventional type heat pipe, which will be the kind used in this design.[2]

Along with being a very good thermal conductor, a heat pipe has the following features:[3]

1.) It is flexible,
2.) It is controllable,
3.) It can act as a thermal diode,
4.) It can act as a heat flux transformer,
5.) It has isothermal surfaces.

A. THEORY

A heat load enters the evaporator region of the hot side heat pipe by means of radiation from being attached to the general purpose heat source (GPHS). The addition of this heat causes the liquid in this section to boil. The resulting vapor flows down the center of the heat pipe to the other end where the condenser heat sink, the hot side of the Stirling engine, is located. As heat leaves the heat pipe the vapor condenses and returns through the wick to the evaporator. This completes a cycle of a hot side heat pipe. The cold side heat pipe's evaporator section of the haeat pipe is attached to the cold side of the Stirling engine and it carries the heat flux to the condenser section. The heat is conducted through a fin plate and dissipated to the Mars atmosphere via thermal radiation. The cold side heat pipes consist of two parts: the actual heat pipe and a radiator. The function of the radiator is to reject the waste heat from the Stirling engine's cold side to the environment (Martian atmosphere). For a heat pipe to operate properly the wick in the evaporator must not be allowed to dryout. This will occur when the maximum capillary pressure head is greater than or equal to the total pressure drop in the heat pipe. The pressure drop depends on the following pressure differentials:

1. the pressure required to return the fluid from the condenser back to the evaporator, ΔP_l,
2. the pressure required to cause the vapor to flow from the evaporator to the condenser, ΔP_v,
3. the pressure head caused by the gravitational field, ΔP_g.

Algebraically, the equation which must be satisfied is:[4]

$$(\Delta P_c)_{max} \geq \Delta P_l + \Delta P_g + \Delta P_v. \tag{3.1}$$

Pressure difference between the vapor and the liquid is caused by surface tension. This results when two fluids are in contact. They exhibit an unbalanced force due to molecular attractions which arise from a tension in the surface of seperation. A pressure difference is present since the pressure of the vapor on the concave side of the surface is less than the pressure of the liquid on the convex side. This concave/convex interface is called the meniscus. The relation between the surface energy, σ_l, and the radius of curvature is written as:[5]

$$\Delta P = \sigma_l (1/R_1 + 1/R_2), \tag{3.2}$$

where R_1, R_2 are radii at right angles to one another.

To picture capillary action, imagine that a straw is placed in a container filled with some liquid. If the liquid wets the material of the straw, the liquid will rise up the straw to a height H above the surface of the liquid. After the fluid has risen to height H the pressure balance is given by:[6]

$$\sigma_l g H = 2\sigma_l/(r\cos\theta), \tag{3.3}$$

where σ_l is the density of the liquid and θ is the angle of contact.

The static pressures of the phases and the local capillary pressure are balanced along the interface between the liquid and the vapor in heat pipes. This state comes about because the interface will adjust its location within the wick such that the pressures are in equilibrium. This equilibruim state is described by:[7]

$$P_v-P_l=2\ \sigma_l\cos\theta/r. \qquad (3.4)$$

If at the end of the condenser there is no pressure difference between the phases maximum circulation will occur.

B. Components

The major design choices for heat pipes are the wick structure, containment structure., and the working fluid. The process of how these are chosen will be briefly discussed, first looking at the working fluid which, in a large part, determines the material that will be used for both the wick and the containment structure.

1. Working Fluid

The selection of the working fluid must be based on thermodynamic considerations which are concerned with the various limitations to heat flow within the heat pipe. These considerations include: viscous, capillary, entrainment,and nucleate boiling limitations. Figure 3.2 graphically display how the limitations occur as a function of increasing operating temperature of the heat pipe.[8] The working fluid in the heat pipes should possess these nine properties:[9]

a. High surface tension in order to provide enough capillary pumping action.

b. Good wetting characteristics.

c. Low viscosity for sufficient circulation rate.

d. Relatively high value for latent heat of vaporization for optimum axial heat transfer.

e. Vapor pressures below a maximum value, determined by the operating.temperature range.

f. High thermal conductivity for maximum heat transfer between the fluid wall and the wick.

g. Freezing and boiling points compatible with the temperature range determined by the design criteria.

h. The working fluid should also be characterized by a high density in order to reduce flow resistance.

Maximum values for the various operating parameters occur when the working fluid is near its boiling point. Since the hot side of the Stirling engine is set at 1000K, the working fluid was chosen to be sodium. Sodium has a normal boiling point temperature of 1156K at atmospheric pressure. The operating temperature at the cold side of the Stirling engine is from 200 to 500K. The working fluids used for this temperature range are freon, ammonia, alcohol, acetone,water, and some organic compound. Water, a widely used medium, with good thermophysical properties as well as operating in the desired temperature range will be chosen as the working fluid for the cold side heat pipes. Other parameters associated with sodium at the temperature set by the hot side of Stirling engine are listed in section 5.A. Table 3.1

lists different working fluids along with there normal operating temperature ranges.[10]

2. Wick Structure

The main purpose of the wick structure is to provide the necessary capillary pressure to circulate the fluid in a cyclic nature required for normal operation of the heat pipe. The wick should also be able to distribut the liquid to all areas which receive heat from a heat source. The other functions of the wick include: providing the flow passage for the return of the condensed liquid, and also to provide a heat flow path between the inner wall of the container and liquid-vapor interface. An effective heat pipe which meets these functions is described as having small surface pores, large internal pores, and an uninterrupted heat flow path across the wick thickness. Decreasing the size of the surface pores allows for greater capillary pressure while not significantly increasing flow resistance in the channels.

The most common wick structure is the wrapped-screen wick. This will be the chosen wick structure for the hot side heat pipes because of past reliable performances in space applications. From compatibility data in Table 3.2[11] and Table 3.3[12] for low temperature working fluids, the wick material for the cold side heat pipes is chosen to be Copper 60. The number of pores per inch is inversely proportional to the surface pore size of the wick. An important characteristic of the wrapped-screen wick is that the wrapping tightness controls the flow resistance of liquid flow. This implies a greater degree of operating flexibility. Increasing the thickness of the porous wick will cause an increase in the heat flux of the entire length of the heat pipe while decreasing the allowable maximum evaporator heat flux. At the same time, the intrinsic, radial thermal resistance of the heat pipe is increased. For space applications the wick structure is restricted by size and required power load capability to having a small pore size in order to optimize the axial liquid flow. Taking the above considerations of the wick structure and the chosen working fluid's temperature range into account, a stainless steel mesh will be chosen for the hot source heat pipes. The reason for choosing stainless steel mesh is because it is considered to be a compatable material with sodium as the working fluid.[13]

3. Container

The container needs to be leak-proof in order to isolate the working fluid from the outside environment. It also needs to keep a pressure differential across its walls while also making it possible to transfer heat into and out of the working fluid. Container material is dependent upon the following characteristics:[14]

a. Compatability with both the surroundings and the working fluid,

b. Good strength to weight ratio,

c. High thermal conductivity to insure a minimal temperature drop between heat source and wick structure,

d. Ease of fabrication,

e. Wettability (perferrably, $\cos\theta = 1$),

f. Nonporous so that outside gas will not diffuse into the heat pipe.

Life tests of stainless steel heat pipes with sodium as the working fluid conducted at evaporator temperatures of 873K, 923K, and, 973K have been performed with favorable results.[15] Therefore the container material for the hot source heat pipes has been chosen to be stainless steel. By compatibility with the working fluid, H_2O, and having good physical properties, copper is chosen as the container material for the cold side heat pipes.

4. OPERATING PARAMETERS

The beginning of a cycle can be considered to be at the evaporator section where the liquid is heated by the GPHS. The liquid is vaporized and flows in the center of the heat pipe toward the condenser section. At the condenser, the vapor is condensed and the latent heat of vaporization is given up to the hot side of the Stirling engine which acts as a heat sink. The condensed liquid is then brought back to the evaporator section via capillary action. There exists a pressure gradient along the vapor flow passage and also in the wick's liquid flow passages. In order for this cycle to continue the pressure along the liquid side of the liquid-vapor interface must be different along the entire length of the heat pipe. This pressure difference is the cause of the capillary action and is established by the menisci that form at the interface. With the formation of the meniscus at the liquid-vapor interface, the capillary pressure is defined as the difference between the vapor pressure and the liquid pressure, and is labelled as P_c. Capillary pressure is calculated by equation (2.2), where R_1 and R_2 are the principal radii of curvature of the meniscus and σ is the surface tension coefficient of the liquid. The maximum capillary pressure is given by $P_{cm}=(2\sigma)/r_c$, where r_c is the maximum value of $(1/R_1+1/R_2)$. For wire screens the effective capillary radius, r_c, can be found by $r_c=(w+d)/2$, where w is the wire spacing and d is the wire diameter. The effective capillary radius, r_c, can also be found by using the mesh number, N, which is defined as the number of wires per unit length by $r_c=(1/2N)$. The maximum heat transport factor $(QL)_{c,max}$ is calculated by the formula:[16]

$$(QL)_{c,max} = P_{cm}/(F_l+F_v) \quad (3.5)$$

where F_l is the liquid frictional coefficient and F_v is the vapor frictional coefficient.

There exists a maximum capillary pressure for any liquid-wick pair. Therefore, for a dryout situation to be avoided in a continuously operating heat pipe, the required capillary pressure should never exceed the value of the maximum capillary pressure. This restriction is called the capillary limitation. Mathematically, this is stated as:[17]

$$P_{max} - P_{min} = P_{c,r} \sim 2\sigma/r_c - \Delta P_{\perp} - \rho_l g l \sin\theta \qquad (3.6)$$

where θ is the angle of inclination of the heat pipe with respect to the horizontal.

5. Heat Pipe Design

A. Hot Side Heat Pipes

1.) Working Fluid: Sodium at 1000K

Liquid Density, r_l=761.826kg/m^3

Vapor density,r_V=0.07268kg/m^3

Liquid viscosity,m_l=1.873(-4)kg/sec

Vapor viscosity,m_l=2.054(-5)kg/msec

Latent heat of vaporization,L=4.06(6)J/kg

Drag coefficient,$(f_V Re_V)$=16

Liquid thermal conductivity,k_l=60.0W/m-K

Liquid surface tension,s_l=0.1303

2.) Container: 316 Stainless Steel

Outside diameter,d_o=0.0254m

Inside diameter,d_i=0.0221m

Vapor core diameter,d_V=0.0201m

Total length=0.65m

Evaporator length,l_{ev}=0.5m

Condenser length,l_c=0.1m

Adiabatic length,l_{ad}=0.05m

Vapor flow area,$A_V=3.14 d_V^2/4$=3.173(-4)m^2

3.) Wick: 316 Stainless Steel

Effective pore radius,r_c=6.1(-5)m

Vapor hydraulic radius,$r_{h,v}$=9.9(-3)m

Thickness of wrapped screen wire,t_W=0.001m

Screen wire spacing,w=5.95(-5)

Screen wire diameter, $d=6.25(-5)m$

Screen mesh number, $N=8197/m$

4.) Maximum Effective Pumping Pressure, P_{pm}:

Capillary radius, $r_c=1/(2N)=6.1(-5)m$

Maximum capillary pressure, $P_c=2s/r_c=4.272(3)n/m^2$

Axial hydrostatic pressure$=r_l g l sinq=-1.852(3)N/m^2$

(Where q is the angle of inclination of the heat pipe with respect to the horizontal, $q=-90$ degrees.)

Normal hydrostatic pressure: $DP^{\wedge}=r_l g d_v cosq=0$

Maximum Effective Pumping Pressure,

$P_{pm}=P_{cm}-DP^{\wedge}-r_l g l sin\theta=6.124(3)N/m^2$

5.) Liquid frictional coefficient, F_l:

Wick cross-sectional area,

$A_w=3.14(d_i^2-d_v^2)/4=6.629(-5)m^2$

Wick crimping factor, $S=1.05$

Wick porisity, $E=1-3.14SNd/4=0.578$

Wick permeability, $k=d^2E^3/(122(1-E)^2)=3.464(-11)$

Liquid frictional coefficient, F_l

$=m_l/(kA_w r_l L)=26.37N/W\text{-}m^3$

6.) Vapor frictional coefficient, F_v:

Vapor flow area, $A_v=3.173(-4)m^2$

Vapor hydraulic radius, $r_{h,v}=9.9(-3)m$

Drag coefficient, $(f_v Re_v)=16$

Vapor Frictional Coefficient, F_v

$=(f_v Re_v)m_v/(2A_v r_{h,v} r_v L)=1.791(-2)N/W\text{-}m^3$

7.) Vapor Dynamic Coefficient, D_v:

Velocity profile coefficient, $B=1.33$

Velocity dynamic coefficient, $D=B/(A_v^2 r_v L^2)$

$=1.103(-5) N/m^2\text{-sec}$

8.) Gravitational Pressure Gradient: $r_l g \sin q = -2849 N/m^3$

$P_{max}-P_{min}= (F_l+F_v)\ Qdx - D_v Q^2 + r_l g \sin q (x_{max}-x_{min})$

9.) Mass Calculations:

a. Mass of container, $M_C=r_S V_C=0.4874kg$

where $r_S=7990 kg/m^3$

and $V_C=6.156(-5)m^3$

b. Mass of wick Strucure, $M_W=(1-E)r_S V_W=0.1453kg$

where $V_W=A_W l=3.76(-5)m^3$

c. Mass of liquid sodium, $M_l=r_l A_W l=0.0145kg$

d. Total mass of 32 hot source heat pipes

$=32(M_C+M_W+M_l)=20.71kg$

Note: The mass of the sodium gas is neglible therefore is not considered in the above calculation of the total mass.

Conclusion for Hot Side Heat Pipes:

The maximum heat load from the general purpose heat source is approximately 7000 W. From the last section the heat load limitation was found to be 249 W. Therefore, the total number of heat pipes needed to transfer the heat from the general purpose heat source to the hot side of the Stirling engine.is 28. Since the geometries of both these components will allow for a maximum of 32 heat pipes, this will be the amount of hot source heat pipes used in this design. This will allow for four redundant heat pipes to act as safeguards against disabling heat pipe failures that would increase the heat load per heat pipe.

The 32 hot side heat pipes will be split betwen three GPHS assemblies. All the hot source heat pipes will be vertical with the evaporator section below the condenser section. The evaporator section will be attached to the general purpose heat source with the condenser section being embedded in the Stirling engine hot side. The operating temperature of the hot source heat pipes will be 1000K, set by the Stirling engine's hot side.

B. Heat Rejection System

The primary purpose of the heat rejection system is to dispose of the waste heat to the environment, the Martian atmosphere . The heat pipe radiator which has an effective heat conductivity much greater than the best metallic conductors (silver and cooper), is used for this application. The radiator system must also operate for the entire lifetime of the power generating unit. This section is comprised of two parts: heat pipe and radiator. All tables, figures and computer programs are shown in the appendix.

Heat Pipe

The temperature of the cold side of Stirling engine is ranged from 200 to 500^{O}K. Heat pipes that are operated at this range of temperature are in the category called low-temperature heat pipe (LHP).

Design of Heat Pipe.

Knowing the chosen parameters of the heat pipe, water as the working fluid, copper as the wick material and the container, it is necessary to assume a heat pipe thickness and geometry, and then calculate the heat transport to condenser section. This heat transport is constrained by sonic, viscious, capillary .etc.. limitations; if it is less than these limitations, the heat pipe's function is performed satisfactorily.

The dimensions of a standard heat pipe is chosen as follows:

length of heat pipe, l = 1m (l_e=.33m, l_a=.33m, l_c=.34m),

container outer diameter, $d_o = 2.54*10^{-2}$m,

container inner diameter, $d_i = 2.21*10^{-2}$m,

vapor core diameter, $d_v = 1.90*10^{-2}$m.

Assuming homogenuous wick, and constant liquid properties along the pipe, and neglected pressure drop due to vapor flow, the maximum mass flow rate, m_{max} is obtained by[18]:

$$m_{max} = (\rho_l * \sigma_l/\mu_l)*(K*A/l)*(2/r_e - \rho_l*g*l*\sin\Theta/\sigma_l),$$

where μ_l - viscosity of the liquid,

σ_l - surface tension of liquid,

ρ_l - density of liquid,

K - wick permeability,

A - wick cross sectional area,

r_e - effective pore radius,

g - gravitational acceleration acting on the pipe,

l - length of the heat pipe,

Θ - angle above horizontal.

The above equation is used to solve the maximum heat transport of the heat pipe, Q_{max} :

$$Q_{max} = m*L,$$

where L is the latent heat of vaporization.

From the above mass flow rate, the maximum flow rate occurs at $\sin\Theta=0$. It means that the heat pipe is arranged horizontally for maximum flow rate. The computer program is written with various wick area from

$0.6*10^{-4}$ to $2.*10^{-4}$ m^2 vs. various heat pipe length from 0.5 to 1.0 m at temperature of 100 ^0C, boiling temperature of water. The heat transport of the chosen heat pipe is calculated from print-out about 250 watts if the total heat needed to transport for rejection is from 3 to 5 Kw. The heat pipes required are from 12 to 20 units. In order to be conservative of the safety of heat rejection system, the average number of heat pipe required is 16 units. Eight heat pipes will be on each end of the cold side of Stirling engine (for convenience, left and right side).

In order to check the sonic and capillary limits of the pipe, the sonic[19] and capillary[20] equations are obtained by:

$$Q^*/A = \rho_0 V_0 h_{fg}/(2^*(K+1))^{**}1/2,$$

where $K = C_p/C_v = 1.67$,

R - gas constant = $8.314*10^7$ erg/deg mole/mole weight,

V_0- velocity of sound = $(K^*R^*T_0)^{**}1/2$,

ρ_0, T_0 - density and temperature of gas.

At the minimum operating temperature, 20^{0}C, the minimum axial heat flux due to the sonic limitation is $\sim$ 1Kw:

$$Q = L^*\sigma^*\cos\Theta/(8^*l_{eff}{}^*d_l{}^*(\upsilon_l/(A_l{}^*d_l{}^{**}2)+(\upsilon_v/(A_v{}^*d_v{}^{**}2)),$$

where l_{eff} - effective heat pipe length,

d_l - hydraulic wick diameter,

d_v - diameter of the vapor duct,

υ_l, υ_v - kinematic viscosity of liquid, gas,

A_l, A_v - wick and vapour area.

This capillary equation is used without taking the effect of buck forces into account because of the hozizontal heat pipe. At the boiling temperature of $100^{o}C$, the heat flux due to the capillary limitation is 5.7Kw. Both of these limits exceed the heat transport of heat pipe, 250 watts. Hence the above heat pipe will operate satisfactorily at temperature range of $300\text{-}500^{o}K$.

The mass of heat pipe is composed of three parts: container, wick and fluid volume.

The mass of container is expressed by:

$$M_c = \rho*\pi(D_o{**}2 - D_i{**}2)/4$$

and mass of wick by

$$M_w = \rho*\pi*(D_i{**}2 - D_v{**}2)*l*(1-\varepsilon)/4$$

and mass of fluid[21] by

$$M_f = \rho*(0.001*D_i*(l_a+l_c) + A_w*l_e*\varepsilon)$$

where ε - wick porosity,

ρ - density associated with material,

other nomenclatures explained in previous equations.

The mass of one heat pipe is found to be: container, 1.1kg, wick, 0.334kg, and fluid, 0.035kg. Therefore, the total weight of 16 heat pipes is found about $\sim$ 23.504 kg.

Radiator

One of the main purposes for a radiator is to maintain a passive system. The good thing about heat pipe radiator is the heat pipe works independently. If one heat pipe is damaged by some reasons, the other heat pipes still function very well. Radiator should be simple and convenient with the choice of either flat plate or attached fins. Fins provide a large area for radiative heat transfer with a relatively small mass. But the view factor between fins reduces their efficiences as well as the fins will accumulate duct that resists the heat radiating into the environment. Winds at 20mph are very common on Mars. On the other hand, a flat plate has more mass associated with it per the amount of heat transfer. The extra weight that is incured by that flat plate system is offset by the reduced view factor and the reduced duct accumulation. By these reasons, a simple but effective flat plate designed heat radiator is chosen. The flat plate is welded to the condender section of heat pipe in making an array as illustrated in figure 3.3.

The means of heat transfer are by way of thermal convection and of radiation. If both of factors are taken into account, the area of heat transfer will be less resulting in smaller mass. But there is no data available for thermal convection coefficient, and for the safety of operating the unmanned vehicle on Mars during a period of 5-10 years, only thermal radiation is taken in account and is characterized by the Stefen-Boltzman equation below[22]:

$Q = \varepsilon*\sigma*A*(T_s**4 - T_{surr}**4)$,

where ε - emissivity,

σ - Stefen-Boltzman constant = $5.67*10^{-8} w/m^2\ ^0K^4$,

A - heat transfer area,

T_s, T_{surr} - surface, surrounding temperature,

Q - heat transfer.

The surrounding temperature is taken as average temperature on Mars. The material chosen for radiator is aluminum because of high thermal conductivity and low density. To increase the emissivity the radiative side of the aluminum sheet, a coating of aluminum oxide, Al_2O_3 will be applied. This coating has en emissivity of 0.85-0.95. With a heat rejection of about 3Kw, and at operating temperature about $500^{o}K$, the area required for heat transfer is about $1 m^2$.

The mass of radiator is expressed by equation below:

$M_r = A*t*\rho$,

where t - thickness.

The thickness size chosen is 0.25 cm. This thickness provides enough stability for the structural integrity of the system as well as for protection of the system from damage. With the above assumptions, the mass of flat plate required is about 6.768 kg.

Conclusion for the Cold Side of Stirling engine (Heat Rejection System)

The weight of heat pipe radiator is 30.272 kg. This radiator is designed to operate at the maximum condition to assure the working function of the system. the system is unmanned vehicle on planet Mars ; no man labor is available when the vehicle is broken down.

The heat pipes are arranged horizontally. The evaporator section will be embebded in the Stirling engine cold side with the condenser section being attached to the flat plate radiator. The plate radiator projecting out of the vehicle like wings of an aircraft is used to reject the waste heat. The operating temperature of the radiator will be from 300 to $500^{o}K$, set by the cold side of Stirling engine.

```
C============================================C
C               . Program Pipe mass flow       C
C        This program is used the maximum mass  C
C        flow rate of the horizontal heat pipe  C
C        without arteries and with a homogenous C
C        wick.                                  C
C        The equation is given by:              C
C         m = (rho*sigma/u)*(k*a/l)*(2/rc-rho*mgC
C              *l/sigma)                        C
C         and the heat transport is:            C
C             Q = ml                            C
C        The range of pipe length 0.5 -- 1.0 m. C
C        The wick cross sectional area 1.E-04 - C
C        2.E-04 meter square.                   C
C                 Nomenclature                  C
C        a      :  area of wick cross sectional  C
C        l      :  length of heat pipe          C
C        rho    :  density of liquid            C
C        sigma :  surface tension of liquid     C
C        u      :  viscosity of liquid          C
C        rc     :  effective pore radius        C
C      lat    :  latent heat                  C
C        k      :  wick permeability            C
C      mg     :  mars gravity                 C
C        q      :  heat flux                    C
C        m      :  mass flow rate               C
C============================================C
      real a,l,q,temp,m
      real lat,mg,rc,rho,sigma,u,k
      open (unit=7,file='ne410.dat',status='new')
      write(6,*)'mars gravity=?'
      read(5,*) mg
      write(6,*) 'temp=?'
      read(5,*) temp
      if (temp.le.0.0) go to 100
      write(6,*)'Rho,Sigma,U,K,Rc,Latent heat=?'
      read(5,*) rho,sigma,u,k,rc,lat
      write (7,5) temp
5     format(/1x,'temp.=',f6.1,'deg. C')
      write(7,6)
6     format(1x,'wick area(m**2)',2x,'pipe length(m)',2x,'mass(kg)',
     *2x,'q(kw)')
20    write (6,*) 'wick area=?'
      read (5,*) a
      write (6,*) 'pipe length=?'
      read (5,*) l
33    m=(rho*sigma/u)*(k*a/l)*(2/rc)
      q=m*lat
      write(7,35) a,l,m,q
35    format(1x,e10.4,4x,f6.4,4x,e10.4,4x,e10.4)
        l=l+0.1
      if (l .le. 1.1) go to 33
        l=l-0.6
        a=a+2.E-05
        if (a .le. 2.2E-04) go to 33
100   END
```

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temp.= 100.0deg. C

wick area(m**2)	pipe length(m)	mass(kg)	q(kw)
0.6000E-04	0.4000	0.1665E-03	0.3760E+00
0.6000E-04	0.5000	0.1332E-03	0.3008E+00
0.6000E-04	0.6000	0.1110E-03	0.2506E+00
0.6000E-04	0.7000	0.9514E-04	0.2148E+00
0.6000E-04	0.8000	0.8325E-04	0.1880E+00
0.6000E-04	0.9000	0.7400E-04	0.1671E+00
0.6000E-04	1.0000	0.6660E-04	0.1504E+00
0.8000E-04	0.5000	0.1776E-03	0.4010E+00
0.8000E-04	0.6000	0.1480E-03	0.3342E+00
0.8000E-04	0.7000	0.1269E-03	0.2864E+00
0.8000E-04	0.8000	0.1110E-03	0.2506E+00
0.8000E-04	0.9000	0.9867E-04	0.2228E+00
0.8000E-04	1.0000	0.8880E-04	0.2005E+00
0.1000E-03	0.5000	0.2220E-03	0.5013E+00
0.1000E-03	0.6000	0.1850E-03	0.4177E+00
0.1000E-03	0.7000	0.1586E-03	0.3581E+00
0.1000E-03	0.8000	0.1388E-03	0.3133E+00
0.1000E-03	0.9000	0.1233E-03	0.2785E+00
0.1000E-03	1.0000	0.1110E-03	0.2506E+00
0.1200E-03	0.5000	0.2664E-03	0.6015E+00
0.1200E-03	0.6000	0.2220E-03	0.5013E+00
0.1200E-03	0.7000	0.1903E-03	0.4297E+00
0.1200E-03	0.8000	0.1665E-03	0.3760E+00
0.1200E-03	0.9000	0.1480E-03	0.3342E+00
0.1200E-03	1.0000	0.1332E-03	0.3008E+00
0.1400E-03	0.5000	0.3108E-03	0.7018E+00
0.1400E-03	0.6000	0.2590E-03	0.5848E+00
0.1400E-03	0.7000	0.2220E-03	0.5013E+00
0.1400E-03	0.8000	0.1943E-03	0.4386E+00
0.1400E-03	0.9000	0.1727E-03	0.3899E+00
0.1400E-03	1.0000	0.1554E-03	0.3509E+00
0.1600E-03	0.5000	0.3552E-03	0.8021E+00
0.1600E-03	0.6000	0.2960E-03	0.6684E+00
0.1600E-03	0.7000	0.2537E-03	0.5729E+00
0.1600E-03	0.8000	0.2220E-03	0.5013E+00
0.1600E-03	0.9000	0.1973E-03	0.4456E+00
0.1600E-03	1.0000	0.1776E-03	0.4010E+00
0.1800E-03	0.5000	0.3996E-03	0.9023E+00
0.1800E-03	0.6000	0.3330E-03	0.7519E+00
0.1800E-03	0.7000	0.2854E-03	0.6445E+00
0.1800E-03	0.8000	0.2498E-03	0.5639E+00
0.1800E-03	0.9000	0.2220E-03	0.5013E+00
0.1800E-03	1.0000	0.1998E-03	0.4512E+00
0.2000E-03	0.5000	0.4440E-03	0.1003E+01
0.2000E-03	0.6000	0.3700E-03	0.8355E+00
0.2000E-03	0.7000	0.3171E-03	0.7161E+00
0.2000E-03	0.8000	0.2775E-03	0.6266E+00
0.2000E-03	0.9000	0.2467E-03	0.5570E+00
0.2000E-03	1.0000	0.2220E-03	0.5013E+00

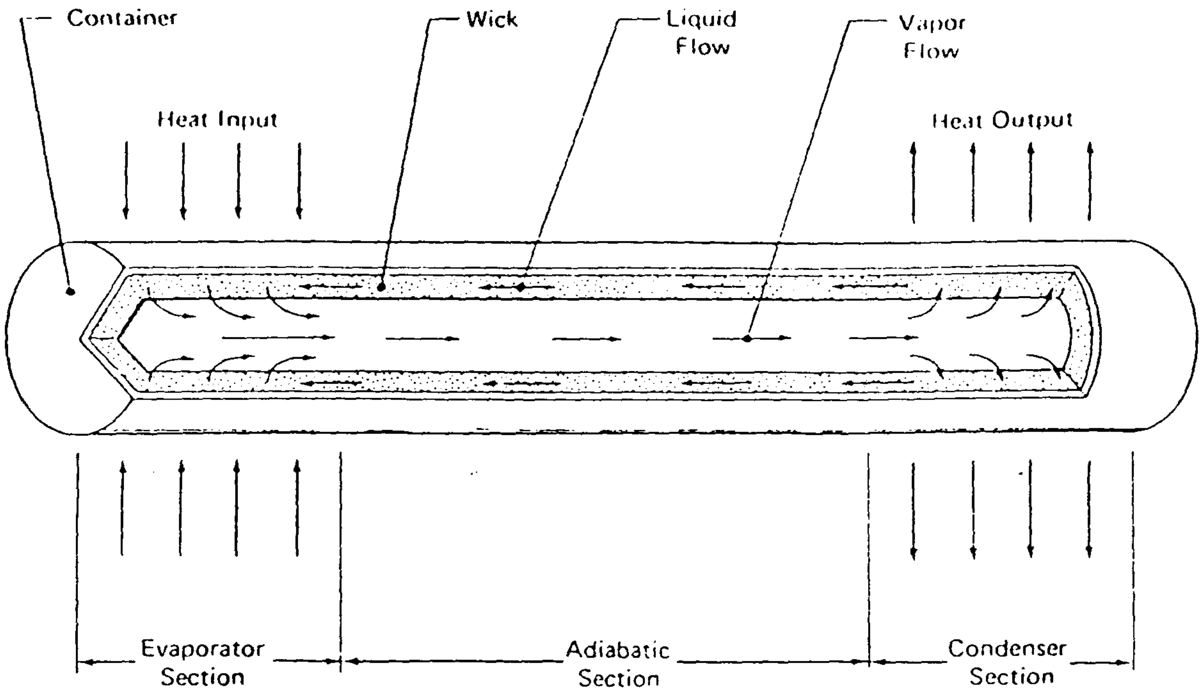

FIG.3-1 Components and principle of operation of a conventional heat pipe.

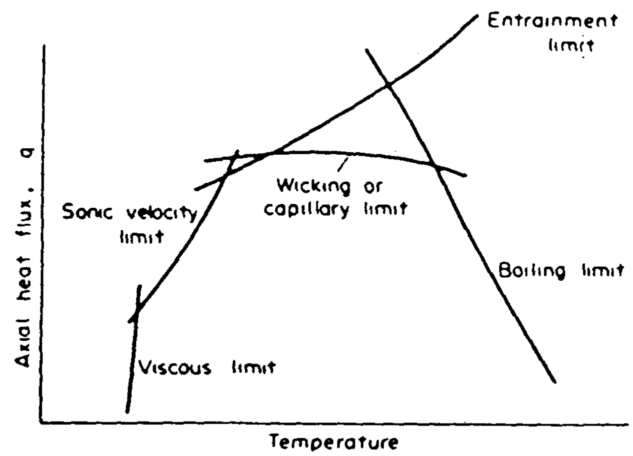

Fig.3.2 Limitations to heat transport in the heat pipe

TABLE 3.1 HEAT PIPE WORKING FLUIDS

(The useful operating temperature range is indicative only). Full properties of most of the below are given in Appendix 1.

Medium	Melting point (°C)	Boiling point at atmos.press. (°C)	Useful range (°C)
Helium	-272	-269	-271 - -269
Nitrogen	-210	-196	-203 - -160
Ammonia	-78	-33	-60 - 100
Freon 11	-111	24	-40 - 120
Pentane	-130	28	-20 - 120
Freon 113	-35	48	-10 - 100
Acetone	-95	57	0 - 120
Methanol	-98	64	10 - 130
Flutec PP2*	-50	76	10 - 160
Ethanol	-112	78	0 - 130
Heptane	-90	98	0 - 150
Water	0	100	30 - 200
Toluene	-95	110	50 - 200
Flutec PP9*	-70	160	0 - 225
Thermex**	12	257	150 - 395
Mercury	-39	361	250 - 650
Caesium	29	670	450 - 900
Potassium	62	774	500 - 1000
Sodium	98	892	600 - 1200
Lithium	179	1340	1000 - 1800
Silver	960	2212	1800 - 2300

*Included for cases where electrical insulation is a requirement.

**Also known as Dowtherm A, an eutectic mixture of diphenyl ether and diphenyl.

TABLE 32 COMPATIBILITY DATA
(Low temperature working fluids)

Wick Material	Working Fluids					
	Water	Acetone	Ammonia	Methanol	Dow-A	Dow-E
Copper	RU	RU	NU	RU	RU	RU
Aluminium	GNC	RL	RU	NR	UK	NR
Stainless Steel	GNT	PC	RU	GNT	RU	RU
Nickel	PC	PC	RU	RL	RU	RL
Refrasil Fibre	RU	RU	RU	RU	RU	RU

RU Recommended by past successful usage

RL Recommended by literature

PC Probably compatible

NR Not recommended

UK Unknown

GNC Generation of gas at all temperatures

GNT Generation of gas at elevated temperatures, when oxide present.

TABLE 33 HUGHES AIRCRAFT COMPATIBILITY RECOMMENDATIONS

	Recommended	Not Recommended
Ammonia	Aluminium	Copper
	Carbon Steel	
	Nickel	
	Stainless Steel	
Acetone	Copper	
	Silica	
	Aluminium*	
	Stainless Steel*	
Methanol	Copper	Aluminium
	Stainless Steel	
	Silica	
Water	Copper	
	Monel	
	347 Stainless Steel **	Stainless Steel
		Aluminium
		Silica
		Inconel
		Nickel
		Carbon Steel
Dowtherm A (Thermex)	Copper	
	Silica	
	Stainless Steel ***	
Potassium	Stainless Steel	Titanium
	Inconel	
Sodium	Stainless Steel	Titanium
	Inconel	

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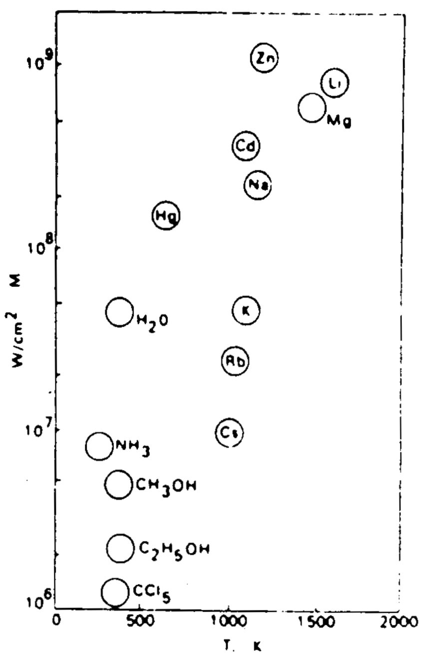

Fig. 3.3 Merit Number for selected working fluids for their boiling point (Courtesy Philips Technical Review)

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IV. Results & Conclusions

Results

The General Purpose Heat Source delivers a minimum 7Kw thermal power over the ten year mission lifetime. Because the heat is generated by Pu^{238} decay and temperatures are well below the melting points for the materials used, the GPHS is inherantly 100% reliable. The pressure from the helium produced is also at a low,easily contained level, 160 psi. The total mass of the GPHS, 175 Kg, is 20-25 Kg less than similiar sources used for space applications. Also, note that the mass of all the GPHS modules together, 36.879 Kg, is close to the optimum mass, 35.6 Kg, predicted by the mass optimization procedure.

The Stirling engine / linear alternator arrangement is a light, compact power generation unit, providing a constant 2Kw electric power. The Stirling engine has a demonstrated high reliability, which is enhanced by the choice of working gas, helium, which is inert, and the low operating pressure, 40 bar. The reliability is also enhanced by having only two moving parts. The overall efficiency, 30%, is as good as or better than most Brayton and Rankine isotope power systems. The linear alternator has an efficiency of 90% in the 2Kw range.

Heat pipes are a highly efficient means of heat transport, and a relatively low number of them were required for this system, 48 total. The pipes are also light weight, so the extra pipes for redundancy did not add much mass to the system, an average of about 1 Kg / pipe. Since the power level is low at 2Kw, the radiator area is small, 1m^2. It will be constructed out of aluminum, keeping the mass to a minimum value of about 7 Kg, close to the optimized value of 5.3 Kg.

The system has a total mass of 175 Kg, which is a weight savings of about 30 Kg over Rankine and Brayton power systems presently in use.

Conclusion

This report describes a feasible power supply system for a sample and data collection vehicle. The system is lower mass than comparable systems, and has a high reliability. System operation is independent of its orientation, a very desirable characteristic. These features combine to make it an attractive choice for unmanned planetary exploration.

V. Mass Optimization

Mass Optimization

A mass optimization for the entire system was performed using relationships between Carnot efficiency and system mass. For the radiator the mass temperature relation used was

$$m_{rad} = Q * t * \rho / [\varepsilon * \sigma * (T_s^4 - T_{surr}^4)]$$

where Q is the heat rejected

t is the thickness

ρ is the density of the radiator

ε is the emissivity

σ is Stephans constant

We must now relate Q to temperature and engine efficiency. The heat rejected will be related to the efficiency of the engine by the equation

$$Q_{rej} = 2000 / (1-\eta)$$

where $$\eta = (1-TC/TH) * 0.58$$

The heat source was modeled using a linear function due to the nature of the specific heat generation. It was found that

$$m_{hs} = [2000/(1-TC/TH) * .58] /193.3$$

where mass is in Kg.

The mass of the Stirling engine and that of the heat pipes was assumed constant for this first iteration in estimation of mass. It was felt that a simple increase in pressure would bring about higher power levels with the same mass. The total mass of the radiator and heat source was therefore

$$m_{tot} = m_{rad} + m_{hs}.$$

The hot side temperature was set to 1000K and TC was varied between the average temperature of the martian surface and 1000K. This optimization is shown in figure 5-1 and gives a minimum mass at 498K. This leads to an engine efficiency of 29.6% with a heat source mass of 35.6Kg and a radiator mass of 5.322 Kg. NASA has also performed an analysis of mass optimization for a linear alternator Stirling engine assembly and also found a temperature ratio of 2.

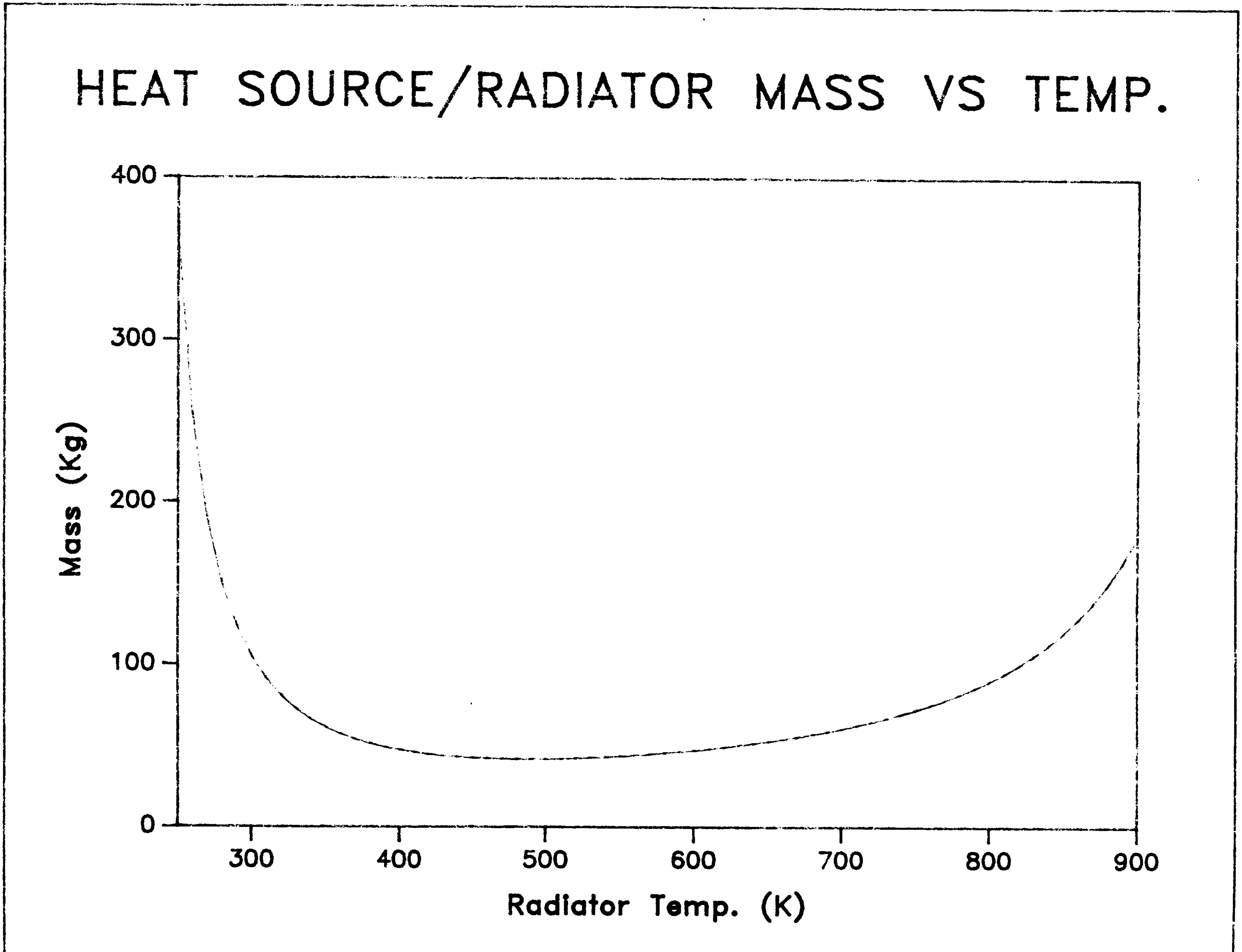

HEAT SOURCE/RADIATOR MASS VS TEMP.
400
300
200
100
0
Mass (Kg)
300
400
500
600
700
800
900
Radiator Temp. (K)

Figure 5-1

Hear Source / Radiator vs Temperature

Optimization Program

```
        real m, x1, x2, x3, ts
        open ( unit=6, file='sysm.dat', status='new' )
c       m = total mass heat source + radiator
c       x1, x2, x3 are temporary variables
c       ts is the cold side temperature
100     x1=1/(1-(1-ts/1000)*0.58)*(1.323*10**8)*2000
        x2=(ts**4-2.5629*10**9)
        x3=2000/((1-ts/1000)*0.58)
        m=x1/x2+x3/193.3
        write (6,105) ts, m
105     format (1x, f5.4, 4x, e10.4 )
        ts=ts+10
        if ( ts.le. 900 ) go to 100
        end
```

VI. Summary

Summary

The power supply design is illustrated in Figure 5-1. The hot side heat pipes run vertically out of the GPHS assemblies and are attached to the Stirling engine/ linear alternator configuration. The cold side heat pipes radiate horizontally from the engine, passing through the vehicle body. The radiator is attached to both the heat pipes and the vehicle body. Note that power supply operation is independent of engine orientation, i.e., it will function horizontally, vertically, upside-down, etc. Figure 5-2 shows the temperature-entropy states for the Stirling engine. One possible vehicle design, Figure 5-3, shows the radiator location and approximate placement of the power supply within the vehicle. Also note the wheel design on the conceptual drawing. The three wheels, mounted tripod fashion and able to rotate, allow the vehicle to move even if it is rolled over onto its top.

Following the figures is an overview of the design specifications for the heat source, heat pipes, and Stirling engine.

System Design

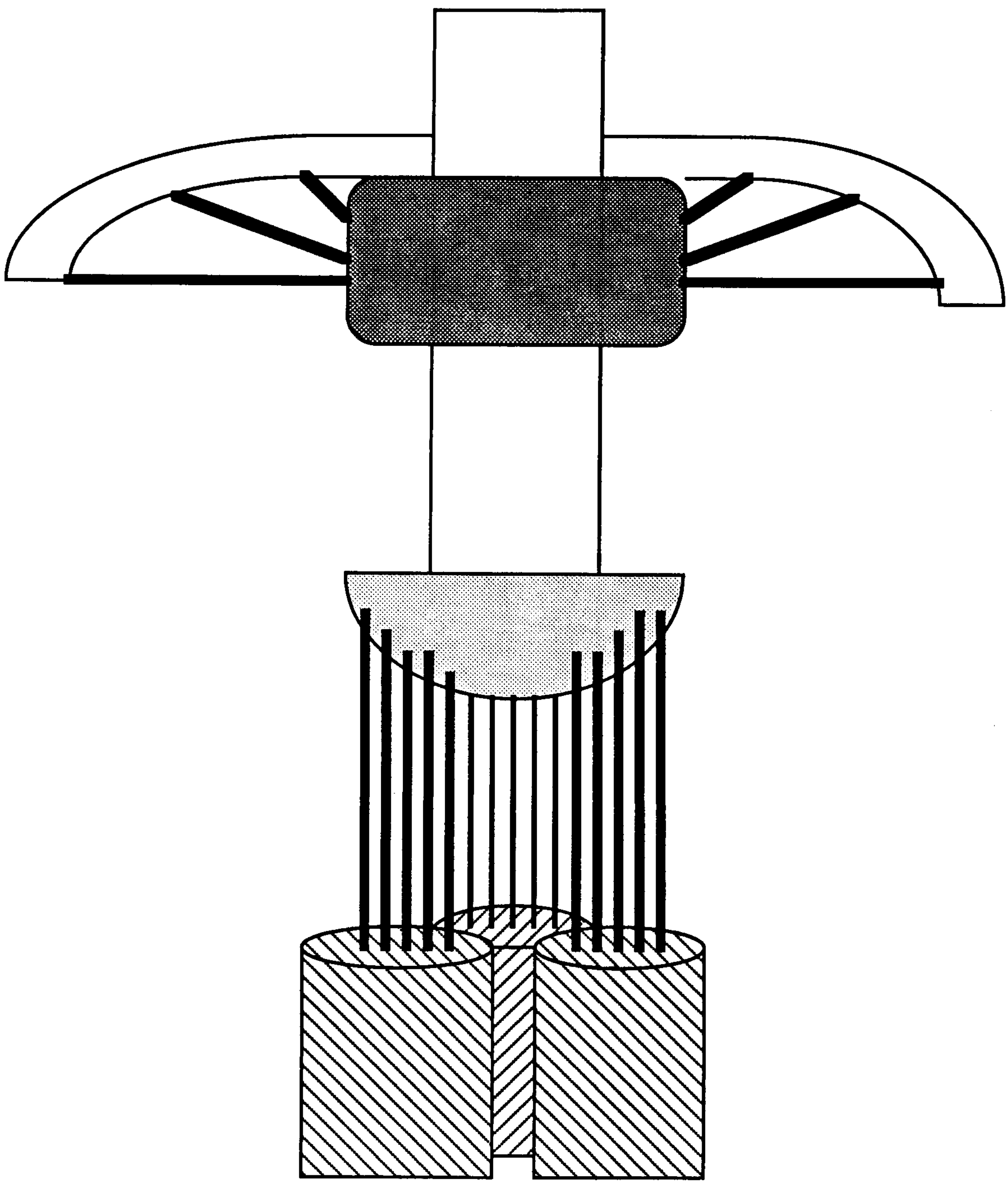
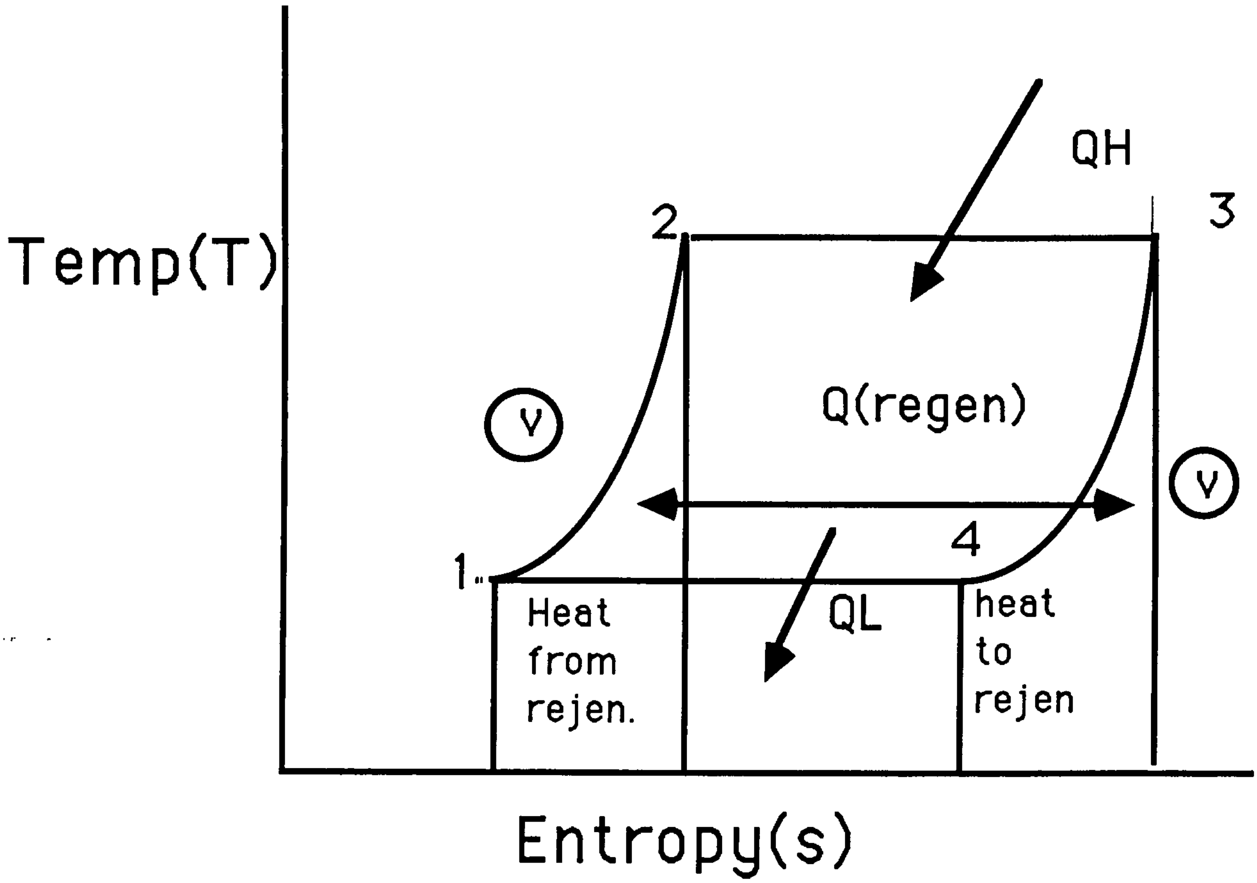


Figure 6-1

Stirling Engine T-S Diagram



V Means Constant Volume Process

1-2 Constant Volume Process
2-3 Isothermal Expansion
3-4 Constant Volume Cooling
4-1 Isothermal Compression

Figure 6-2

Possible Vehicle Design

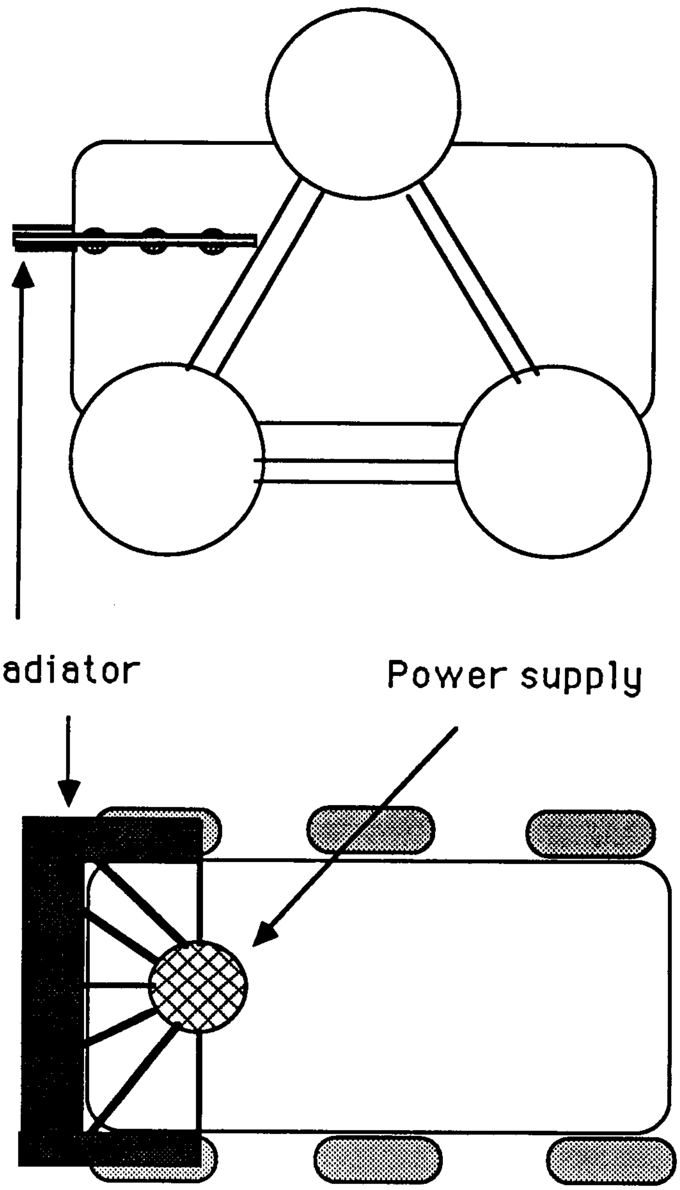


Figure 6-3

GPHS Module

Dimensions: 10.16 cm x 10.16 cm x 5.08 cm

Fuel: PuO_2, 4 cylindrical pellets

Diameter = Height = 2.52 cm

Power: 250 W_{th}

Mass: 1.293 Kg.

GPHS Assembly

10 Modules / Assembly

Casing: Titanium, 0.5 cm thick

Height = 50 cm, Diameter = 20 cm

Mass = 8.638 Kg.

Helium pressure build-up: 160 psi after 10 years.

Maximum centerline temperature: 1775 K

Total Assembly Mass = 21.567 Kg.

Entire Heat Source Assembly

Three GPHS Assemblies

Total Mass = 64.704 Kg.

Power: 7.5 Kw (initially)

6.92 Kw (after 10 years)

Hot Side Heat Pipes

Material: Stainless Steel (316)

Diameter: 2.54 cm

Length: evaporator: 50 cm

adiabatic : 5 cm

condenser : 10 cm

total length: 65 cm

Wick material: Stainless Steel (316)

Working fluid : Sodium

Operating temperature: 1000K

Maximum capillary pressure: 0.042 MPa = 6.076 psi

Mass: pipe : 0.487 Kg

wick: 0.145 Kg

fluid: 0.0145 Kg

Total Mass (32 pipes): 20.714 Kg

Cold Side Heat Pipes

Material: copper

Diameter: 2.54 cm

Length: evaporator: 33 cm

adiabatic : 33 cm

condenser : 34 cm

total length: 1 m

Wick material: copper-60

Working fluid: water

Operating temperature: 500K

Mass: pipe: 1.1 Kg

wick: 0.334 Kg

fluid: 0.035 Kg

Total mass (16 pipes) : 23.504 Kg

Radiator

Material: aluminum

Area : 1 m^2

Mass : 6.768 Kg

Coating : Al_2O_3

Stirling Engine

Material: housing: beryllium

engine : stainless steel and aluminum

Engine mass : 60 Kg

Engine dimensions: length: 60 cm

diameter: 20 cm

Engine displacement: 93.3 cm^3

Engine efficiency : 29.6%

Regenerator efficiency: 62%

Hot side temperature: 1000K

Cold side temperature: 500K

Working gas: helium

Operating pressure: 40 bar

Power output (from linear alternator): 2Kw electric

Total System

System mass: 175.69 Kg

System dimensions: height: 125 cm

width : 50 cm

(excluding cold side pipes/radiator)